Solutions Manual
for
Introduction to Internal Combustion Engines

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Springer

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PREFACE

The popularity of *Introduction to Internal Combustion Engines* during the last ten years or so, has prompted me to prepare this set of worked solutions for the *3rd Edition*. I have used many of the questions as exam questions, so during the marking of the scripts, I hope that I will have removed any errors. None the less, readers may encounter some errors, in which case I would be very grateful to hear about them.

So as to make this manual self-contained, I have included the questions, and for clarity these have been written **in bold type**. The questions in the 2nd Edition have been denoted by a superscript asterisk (*). At the end of many questions I have added a **Discussion**. This contains material that is strictly speaking beyond the scope of the question, but is intended to be useful supplementary information. This manual provides a useful means of adding supplementary material, as occasionally some of the discussion material, (especially in the discussion-type questions) is not in the book.

All Figure numbers and Equation numbers refer to the *3rd Edition* of the book, and to avoid any confusion, letters have been used in this *Manual* to define equations. The nomenclature and abbreviations are consistent with the book, with the exception of time derivatives for which it has not been possible to include the 'dot'. This means, for example, that work and power are both denoted by *W*, but it should be clear from the comments, units and context what is being used. ***Bold italics*** are used on occasion to indicate entries in the index for topics that are treated in the book, and do not form a core part of the solution. The final answers to each part of a question are shown in **bold font**.

In order to make this manual self contained, the Appendix contains thermodynamic tables for combustion calculations; this should also ensure consistency in the answers if these tables are made available to students. The tables contain equilibrium constants, and molar thermodynamic property data (internal energy, enthalpy, Gibbs Energy, and Entropy) for the species to be found in the reactants and products. The datum for enthalpy adopted here, is a datum of ***zero enthalpy for elements when they are in their standard state at a temperature of 25°C***. The enthalpy of any molecule at 25°C will thus correspond to its enthalpy of formation, $\Delta H_f°$. This choice of datum (although not used to my knowledge in any other tables) will be seen to facilitate energy balances in combustion. Unlike tables that use an identical datum for reactant and product species, there is no need to include enthalpies or internal energies of reaction. Since the first publication of *Introduction to Internal Combustion Engines* the preferred practice for denoting molar quantities is to use a lower case letter (to emphasise that it is a specific quantity) with a tilde above, for example ñ. However, for consistency with the book (and in common with many other publications) molar quantities will be denoted here by upper case letters. This requires the reader to decide whether the symbol refers to a molar specific quantity, or a property value of the complete system.

Some of the combustion problems have been generated using the equilibrium solving package STANJAN, developed by Prof W C Reynolds of Stanford University.

Richard Stone — Summer 1999

CONTENTS

1 Introduction

1.1 What are the key technological developments for spark ignition engines in the first half of the 20th century.

1) Ricardo 1919 Patent for (side valve) turbulent combustion chamber. With the comparatively low octane rating fuel available for normal use (say 60-70), the compression ratio was limited to about 4:1. The Ricardo chamber allowed an increase in compression ratio to 5:1, and when this was combined with the faster burn, there was about a 20% improvement in both the output and the efficiency.

2) Work by Ricardo (for Shell) showed in 1917 that gasoline distilled from Borneo crude oil contained high levels of aromatics (notably: benzene, toluene and xylene). This fuel had a higher octane rating that permitted a unity increase in the compression ratio, with a 10% increase in power and slightly greater reduction in the fuel consumption. It was this fuel that enabled Alcock and Brown to cross the Atlantic in 1919 - they used a modified Rolls Royce Eagle engine, in which the compression ratio was increased from 5:1 to 6:1.

3) In 1922 Midgley and Boyd used tetra ethyl lead [$Pb(C_2H_5)_4$] to increase the octane rating of conventional gasolines (Journal of Industrial and Engineering Chemistry, 1922). This in due course led to the widespread availability of high octane rating fuels (above say 90) with the possibility of compression ratios of 8:1 and higher. This led to the demise of side valve engines since this type of geometry was limited to compression ratios in the region of 5:1. The only sensible way to obtain high compression ratios was to have the combustion chamber above the piston and an overhead valve arrangement.

4.5* Contrast high-turbulence, high compression ratio combustion chambers with those designed for lower compression ratios.

The highest useful compression ratio (HUCR) in spark ignition engines is usually defined by the onset of 'knock' - when the unburnt mixture is self-ignited the ensuing rapid pressure rise can cause structural oscillations that are audible as 'knock'. Self-ignition occurs when the unburnt mixture has been at a high enough temperature for sufficient time, and clearly increasing the compression ratio will increase the unburnt mixture temperatures. Knock is undesirable, since the pressure waves in the combustion chamber disrupt the thermal boundary layers, and this can lead to overheating of components. However, if the combustion is sufficiently rapid, then there will be insufficient time for self-ignition to occur, and instead the whole of the unburnt mixture will be consumed by the steady propagation of the turbulent flame front. Figure 4.9 (redrawn here in problem 4.11) shows that for a stoichiometric mixture, then the turbulent combustion system allows the compression ratio to be increased from about 8:1 to 10:1.

Figure 4.9 also shows that knock is most likely to occur with mixtures that are slightly rich of stoichiometric. The more turbulent combustion means that combustion is centred more closely around top centre, when the unburnt mixture temperature is highest, and this extends the weak and rich mixture flammability limits. Thus, when a lean burn strategy is possible, the compression ratio can be further increased.

4.6* Two spark ignition petrol engines having the same swept volume and compression ratio are running at the same speed with wide open throttles. One engine operates on the two-stroke cycle and the other on the four-stroke cycle. State with reasons:

(i) which has the greater power output
(ii) which has the higher efficiency

The two-stroke engine will have the higher power output, because there are twice as many firing strokes per second. However, the power output will not be double, because: the two-stroke engine will have a lower efficiency (see below), and the quantity of charge trapped in the cylinder will be lower (there will be more exhaust residuals (since the two-stroke engine does not have a separate exhaust stroke, and the transfer port in the two-stroke engine is likely to be closed later than the inlet valve in the four-stroke engine.

The two-stroke engine will have a lower efficiency because of: short-circuiting, in which fuel flows directly from the transfer passage to the exhaust port, and the actual compression ratio will be lower because of the later trapping of the in-cylinder charge.

4.7* The Rover M16 spark ignition engine has a swept volume of 2.0 litres, and operates on the four-stroke cycle. When installed in the Rover 800, the operating point for a vehicle speed of 120km/h corresponds to 3669 rpm, and a torque of 71.85 Nm, for which the specific fuel consumption is 298 g/kWh.

Calculate the bmep at this operating point, the arbitrary overall efficiency and the fuel consumption (litres/100km). If the gravimetric air fuel ratio is 20:1, calculate the volumetric efficiency of the engine, and comment on the value.

The calorific value of the fuel is 43 MJ/kg, and its density is 795 kg/m^3. Ambient conditions are 27°C and 1.05 bar.

Explain how both lean-burn engines and engines fitted with three-way catalysts obtain low exhaust emissions. What are the advantages and disadvantages of lean-burn operation?

6 Induction and Exhaust Processes

6.1* Two possible overhead valve combustion chambers are being considered, the first has two valves, and the second design has four valves per cylinder. The diameter of the inlet valve is 23 mm for the first design and 18.5 mm for the second design. If the second design is adopted, show that the total valve perimeter is increased by 60.8 per cent. If the valve lift is restricted to the same fraction of valve diameter, calculate the increase in flow area. What are the additional benefits in using four valves per cylinder?

In the first case with 2 valves (one inlet and one exhaust), the perimeter (P_2) is:

$$P_2 = \pi d = \pi 23 = 72.26 \text{ mm}$$

In the second case with 4 valves (two inlet and two exhaust), the perimeter (P_4) is:

$$P_4 = 2\pi d = 2\pi 18.5 = 116.24 \text{ mm}$$

The percentage increase in valve perimeter is: $100 \times (P_4 - P_2)/P_2 = 100 \times (116.24 - 72.26)/72.26 = 60.86\%$

Using the valve curtain area (A_c)to define the flow area:

$$A_c = P \times L = \pi k d^2, \qquad \text{where the valve lift } (L) \text{ is: } L = kd$$

Thus:

$$A_{c,2} = \pi k d^2 = \pi k 23^2 = 1662k \text{ mm}^2$$

and:

$$A_{c,4} = 2\pi k d^2 = 2\pi k 18.5^2 = 2150k \text{ mm}^2$$

The increase in valve curtain area is: $(A_{c,4} - A_{c,2})/A_{c,2} = (2150 - 1662)/1662 = 0.29$, or 29%

The other advantages of four valves/cylinder are:

1) There will be a lower camshaft loading. The mass of the valves will scale with length cubed, while the contact length on the cam will scale approximately linearly. The force on the cams that arises from accelerating the valves is proportional to the mass multiplied by the acceleration: the acceleration is proportional to the lift squared. Thus the loading on the cam is proportional to the fifth power of the valve size.

2) Two inlet valves gives greater flexibility for controlling the combustion process. The use of two valves facilitates the generation of tumble (also known as barrel swirl), and this is important of generating homogeneous turbulence just prior to combustion. If one of the two inlet valves is disabled at part load operation, then the air velocity will be increased (leading to more tumble) and the asymmetry will add swirl about an axis parallel to the cylinder axis. The higher levels of swirl will enhance the combustion rate, and compensate for the tendency of the combustion to become slower at part load operation. Clearly the pressure drop across the inlet valve(s) will be increased (in order to give the flow more kinetic energy), but this will not matter, since the same mass of air has to be admitted, and there will be a corresponding reduction in the pressure drop across the inlet valve.

6.2* Describe the differences in valve timing on a naturally aspirated Diesel engine, a turbocharged Diesel engine, and a high performance petrol engine?

The naturally aspirated Diesel engine is likely to have valve timings something like:

The isentropic compressor efficiency (η_c) is (refer to Fig 9.29 in Qu 9.1):

$$\eta_c = (T_{4s} - T_3)/(T_4 - T_3)$$

We now need to find the isentropic compression temperature (T_{4s}) from the pressure ratio across the compressor:

$$T_{4s} = T_3\ (p_4/p_3)^{([\gamma-1]/\gamma)} = (63.5 + 273.15) \text{ x } (2.5)^{([1.4 - 1]/1.4)} = 437.4 \text{ K}$$

So $\eta_c = (437.4 - 336.7)/(217 - 63.5) = \mathbf{0.66}$

b) Next we need to find the value of the compressor mass flow parameter (m^*) for the low pressure compressor, taking careful note of the units since the mass flow parameter is not dimensionless:

$$m^* = m_a \text{ (kg/s) x} \sqrt{T} \text{ (K) / } p \text{ (bar)} = 0.7304 \text{ x} \sqrt{298} \text{ / 1} = 12.61$$

We now need to find the equivalent mass flowrate (m') at the compressor map conditions of 15°C and 1 bar:

$$m^* = m' \text{ (kg/s) x} \sqrt{T} \text{ (K) / } p \text{ (bar)} = m' \text{ x} \sqrt{288} \text{ / 1} = 12.61$$

or $m' = 12.61/\sqrt{288} = 0.743$ kg/s

This needs to be expressed as a volume flowrate (at the map datum conditions), so using the equation of state:

$$V' = m'RT/p = 0.743 \text{ x } 287 \text{ x } 288 \text{ / } 10^5 = 614 \text{ L/s}$$

From the compressor map, the equivalent ambient flowrate of 614 L/s with a pressure ratio of 2.14, indicates a compressor isentropic efficiency of 0.63 - this is quite close to the value of 0.68 calculated from the temperatures.

Next we need to find the value of the compressor mass flow parameter (m^*) for the high pressure compressor, taking careful note of the units since the mass flow parameter is not dimensionless:

$$m^* = m_a \text{ (kg/s) x} \sqrt{T} \text{ (K) / } p \text{ (bar)} = 0.7304 \text{ x} \sqrt{337} \text{ / 2.14} = 6.266$$

We now need to find the equivalent mass flowrate (m') at the compressor map conditions of 15°C and 1 bar:

$$m^* = m' \text{ (kg/s) x} \sqrt{T} \text{ (K) / } p \text{ (bar)} = m' \text{ x} \sqrt{288} \text{ / 1} = 6.266$$

or $m' = 6.266/\sqrt{288} = 0.369$

This needs to be expressed as a volume flowrate (at the datum conditions of the map), so using the equation of state:

$$V' = m'RT/p = 0.369 \text{ x } 287 \text{ x } 288 \text{ / } 10^5 = 305 \text{ L/s}$$

From the compressor map, the equivalent ambient flowrate of 305 L/s with a pressure ratio of 2.50, indicates a compressor isentropic efficiency of 0.65 - this is very close to the value of 0.66 calculated from the temperatures.

c) The effectiveness of the low pressure intercooler is given by:

$$\epsilon = (T_2 - T_3)/(T_2 - T_o) = (168 - 63.5)/(168 - 15) = 0.67$$